GAS POWER SYSTEMS

INTRODUCTION TO INTERNAL COMBUSTION ENGINES

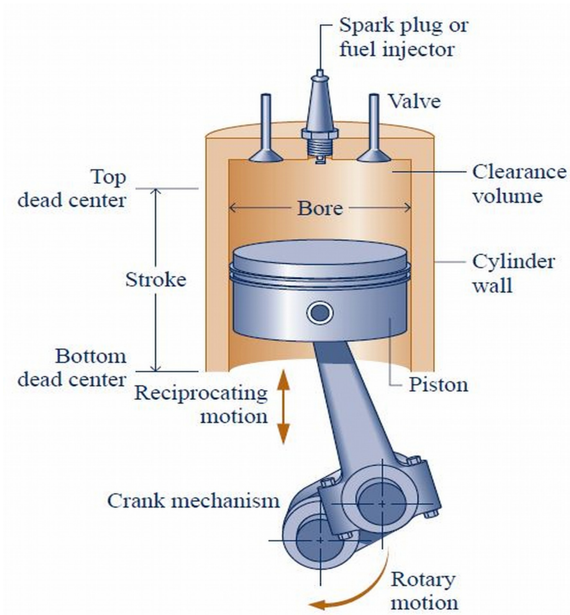
*Two principal types of reciprocating internal combustion engines are the **spark-ignition** engine and the **compression-ignition** engine.*

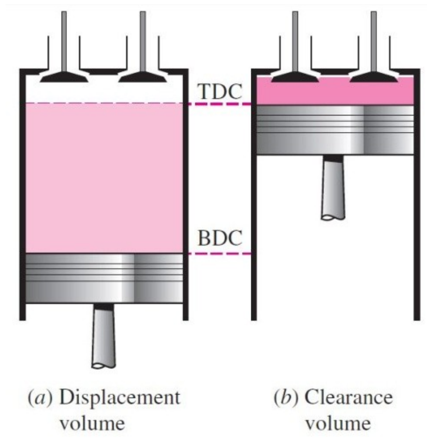
In a spark-ignition engine, a mixture of fuel and air is ignited by a spark plug. Spark-ignition engines have advantages for applications requiring power up to about 225 kW (300 horsepower). Because they are relatively light and lower in cost, spark-ignition engines are particularly suited for use in automobiles.

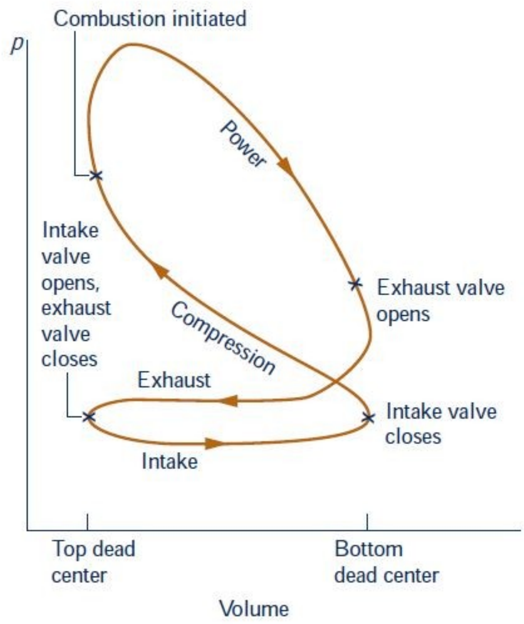
In a compression-ignition engine, air is compressed to a high enough pressure and temperature that combustion occurs spontaneously when fuel is injected. Compression-ignition engines are normally preferred for applications when fuel economy and relatively large amounts of power are required (heavy trucks and buses, locomotives and ships, auxiliary power units).

Nomenclature For Reciprocating Piston Cylinder ICE

The bore of the cylinder is its diameter.
The stroke is the distance the piston moves in one direction.
The piston is said to be at top dead center when it has moved to a position where the cylinder volume is a minimum. This minimum volume is known as the clearance volume. When the piston has moved to the position of maximum cylinder volume, the piston is at bottom dead center. The volume swept out by the piston as it moves from the top dead center to the bottom dead center position is called the displacement volume. In a four-stroke internal combustion engine, the piston executes four distinct strokes within the cylinder for every two revolutions of the crankshaft.







P-V Curve For Reciprocating **ICE**

1) *With the intake valve open, the piston makes an intake stroke to draw a fresh charge into the cylinder. For spark-ignition engines, the charge is a combustible mixture of fuel and air. Air alone is the charge in compression-ignition engines.*

2) *With both valves closed, the piston undergoes a compression stroke, raising the temperature and pressure of the charge. A combustion process is then initiated, resulting in a high-pressure, high-temperature gas mixture. Combustion is induced near the end of the compression stroke in spark-ignition engines by the spark plug. In CI engines, combustion is initiated by injecting fuel into the hot compressed air, beginning near the end of the compression stroke and continuing through the first part of the expansion.*

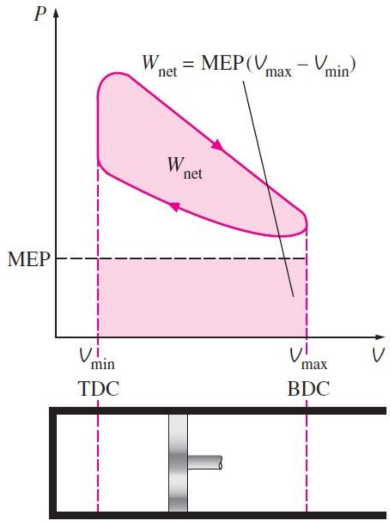
3) *A power stroke follows the compression stroke, during which the gas mixture expands and work is done on the piston as it returns to bottom dead center.*

4) *The piston then executes an exhaust stroke in which the burned gases are purged from the cylinder through the open exhaust valve. The pressure in the cylinder is slightly above the atmospheric value during the exhaust stroke and slightly below during the intake stroke.*

*MEP: The **mean effective pressure** is the theoretical constant pressure that, if it acted on the piston during the power stroke, would produce the same net work as actually developed in one cycle.*

$$\text{mep} = \frac{\text{net work for one cycle}}{\text{displacement volume}}$$

$$\text{MEP} = \frac{W_{net}}{V_{max} - V_{min}} = \frac{w_{net}}{v_{max} - v_{min}}$$



*COMPRESSION RATIO: The **compression ratio** r is defined as the volume at bottom dead center divided by the volume at top dead center.*

$$r = \frac{V_{max}}{V_{min}} = \frac{V_{BDC}}{V_{TDC}}$$

*THERMAL EFFICIENCY: Heat engines are designed for the purpose of converting thermal energy to work, and their performance is expressed in terms of the **thermal efficiency**, which is the ratio of the net work produced by the engine to the total heat input.*

$$\eta_{th} = \frac{W_{net}}{Q_{in}} \quad \text{or} \quad \eta_{th} = \frac{w_{net}}{q_{in}}$$

Thermal efficiency increases with an increase in the average temperature at which heat is supplied to the system or with a decrease in the average temperature at which heat is rejected from the system.

Heat engines that operate on a totally reversible cycle, such as the Carnot cycle, have the highest thermal efficiency of all heat engines operating between the same temperature levels.

The ideal cycles are internally reversible, but, unlike the Carnot cycle, they are not necessarily externally reversible. They may involve irreversibilities external to the system such as heat transfer through a finite temperature . Therefore, the thermal efficiency of an ideal cycle, in general, is less than that of a totally reversible cycle operating between the same temperature limits.

The idealizations and simplifications commonly employed in the analysis of power cycles are:

- *The cycle does not involve any friction. Therefore, the working fluid does not experience any pressure drop as it flows in pipes or devices such as heat exchangers.*
- *All expansion and compression processes take place in a quasiequilibrium manner.*
- *The pipes connecting the various components of a system are well insulated, and heat transfer through them is negligible*

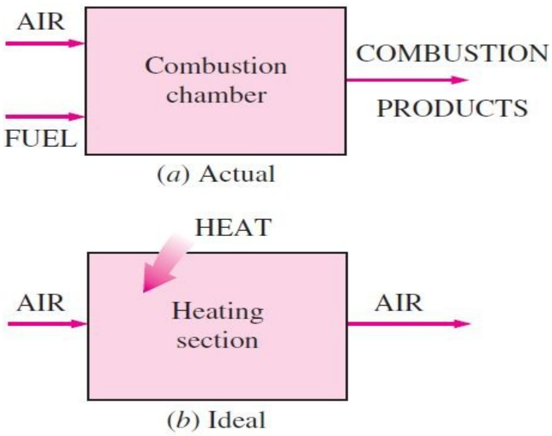
Air-Standard Analysis

Air Standard Assumptions:

- *The working fluid is air, which continuously circulates in a closed loop and always behaves as an ideal gas.*
- *All the processes that make up the cycle are internally reversible.*
- *The combustion process is replaced by a heat-addition process from an external source.*
- *The exhaust process is replaced by a heat-rejection process that restores the working fluid to its initial state.*
- *In a cold air-standard analysis, the specific heats are assumed constant at their ambient temperature values.*

With an air-standard analysis, we avoid dealing with the complexities of the combustion process and the change of composition during combustion.

A detailed study of the performance of a reciprocating internal combustion engine would include the combustion process occurring within the cylinder and the effects of irreversibilities associated with friction and with pressure and temperature gradients. Heat transfer between the gases in the cylinder and the cylinder walls and the work required to charge the cylinder and exhaust the products of combustion also would be considered.



IDEAL GAS MODEL

$$pv = RT$$
$$pV = mRT$$

Changes in u and h:

$$u(T_2) - u(T_1) = \int_{T_1}^{T_2} c_v(T)\, dT$$

$$h(T_2) - h(T_1) = \int_{T_1}^{T_2} c_p(T)\, dT$$

$$u(T_2) - u(T_1) = c_v(T_2 - T_1)$$
$$h(T_2) - h(T_1) = c_p(T_2 - T_1)$$

Changes in s:

$$s(T_2, v_2) - s(T_1, v_1) = \int_{T_1}^{T_2} c_v(T)\frac{dT}{T} + R\ln\frac{v_2}{v_1} \qquad (6.17)$$

$$s(T_2, p_2) - s(T_1, p_1) = \int_{T_1}^{T_2} c_p(T)\frac{dT}{T} - R\ln\frac{p_2}{p_1}$$

$$s(T_2, v_2) - s(T_1, v_1) = c_v \ln\frac{T_2}{T_1} + R\ln\frac{v_2}{v_1}$$

$$s(T_2, p_2) - s(T_1, p_1) = c_p \ln\frac{T_2}{T_1} - R\ln\frac{p_2}{p_1}$$

For variable specific heats:

$$s(T_2, p_2) - s(T_1, p_1) = s^\circ(T_2) - s^\circ(T_1) - R\ln\frac{p_2}{p_1}$$

For reversible adiabatic process/isentropic process-

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{(k-1)/k}$$

$$\frac{T_2}{T_1} = \left(\frac{v_1}{v_2}\right)^{k-1}$$

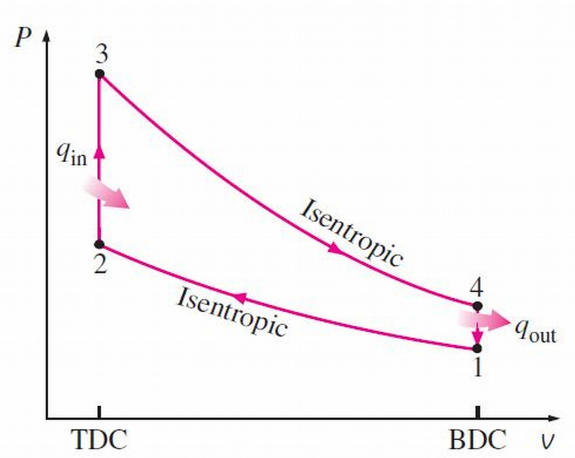
$$\frac{p_2}{p_1} = \left(\frac{v_1}{v_2}\right)^{k}$$

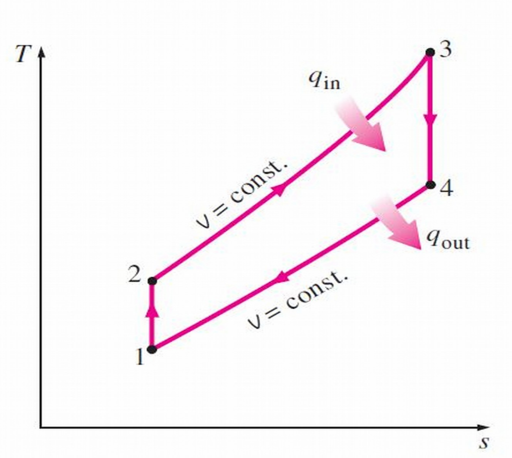
where $k = c_p/c_v$

OTTO-CYCLE

The air-standard Otto cycle is an ideal cycle that assumes heat addition occurs instantaneously while the piston is at top dead center. The Otto cycle is the ideal cycle for spark-ignition reciprocating engines.
It consists of four internally reversible processes:

1-2 Isentropic compression
2-3 Constant-volume heat addition
3-4 Isentropic expansion
4-1 Constant-volume heat rejection





Cycle Analysis:

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = \Delta u$$

$$q_{in} = u_3 - u_2 = c_v(T_3 - T_2)$$

$$q_{out} = u_4 - u_1 = c_v(T_4 - T_1)$$

$$\eta_{th,Otto} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$$

Processes 1-2 and 3-4 are isentropic, and $v_2 = v_3$ and $v_4 = v_1$.

$$\frac{T_1}{T_2} = \left(\frac{v_2}{v_1}\right)^{k-1} = \left(\frac{v_3}{v_4}\right)^{k-1} = \frac{T_4}{T_3}$$

$$r = \frac{V_{max}}{V_{min}} = \frac{V_1}{V_2}$$

$$\eta_{th,Otto} = 1 - \frac{1}{r^{k-1}}$$

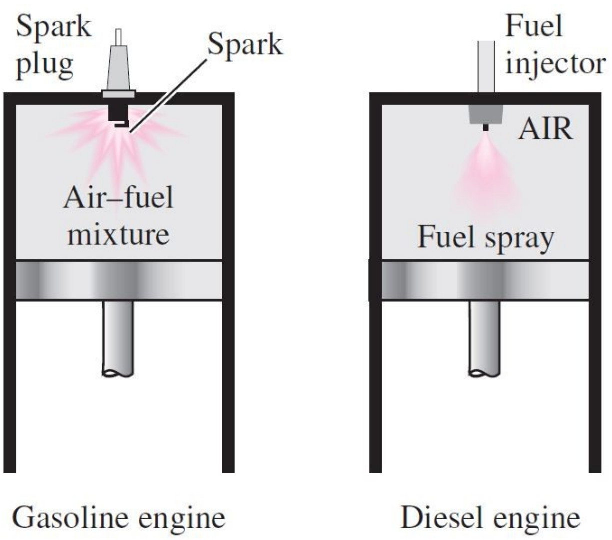
- *The thermal efficiency of the ideal Otto cycle increases with both the compression ratio and the specific heat ratio.*
- *The possibility of autoignition (premature ignition of the fuel), or "knock," places an upper limit on the compression ratio of spark ignition engines. Autoignition may result in high-pressure waves in the cylinder that can lead to loss of power as well as engine damage.*
- *The compression ratios of sparkignition engines are in the range 9.5 to 11.5, approximately. The thermal efficiencies of actual spark-ignition engines range from about 25 to 30 percent.*
- *Otto cycle using a monatomic gas (such as argon or helium, k =1.667) as the working fluid will have the highest thermal efficiency. The specific heat ratio k, and thus the thermal efficiency of the ideal Otto cycle, decreases as the molecules of the working fluid get larger. Specific heat ratio also decreases with temperature.*

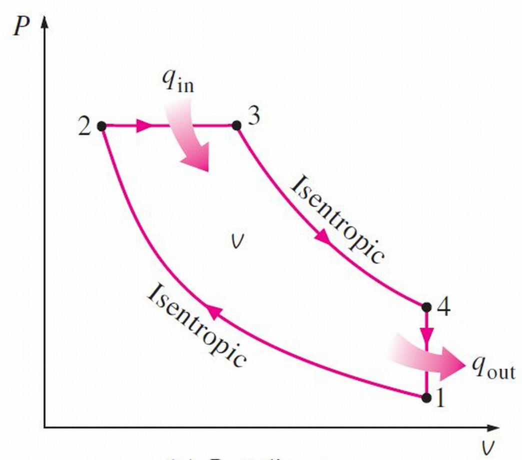
DIESEL CYCLE

The air-standard Diesel cycle is an ideal cycle that assumes heat addition occurs during a constant-pressure process that starts with the piston at top dead center. The Diesel cycle is the ideal cycle for CI reciprocating engines.

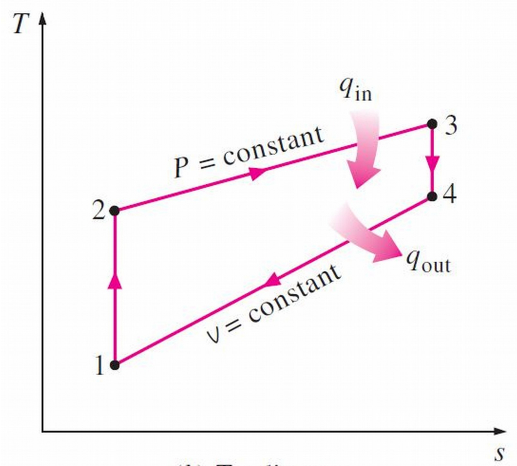
In CI engines (also known as diesel engines),the air is compressed to a temperature that is above the autoignition temperature of the fuel, and combustion starts on contact as the fuel is injected into this hot air. The spark plug and carburetor are replaced by a fuel injector in diesel engines. In diesel engines, only air is compressed during the compression stroke, eliminating the possibility of autoignition. Therefore, diesel engines can be designed to operate at much higher compression ratios, typically between 12 and 24.

The fuel injection process in diesel engines starts when the piston approaches TDC and continues during the first part of the power stroke. The combustion process in these engines takes place over a longer interval; because of this longer duration, the combustion process in the ideal Diesel cycle is approximated as a constant-pressure heat-addition process.





(a) P- v diagram



(b) T-s diagram

Cycle Analysis:

Process 2–3 involves both work (out) and heat (in) ; work(out) is given by-

$$\frac{W_{23}}{m} = \int_2^3 p\, dv = p_2(v_3 - v_2)$$

$$\eta_{th,Otto} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{T_3 - T_2} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$$

$$-q_{out} = u_1 - u_4 \rightarrow q_{out} = u_4 - u_1 = c_v(T_4 - T_1)$$

$$\eta_{th,Diesel} = \frac{w_{net}}{q_{in}} = 1 - \frac{q_{out}}{q_{in}} = 1 - \frac{T_4 - T_1}{k(T_3 - T_2)} = 1 - \frac{T_1(T_4/T_1 - 1)}{kT_2(T_3/T_2 - 1)}$$

$$p_3 = p_2$$

$$V_4 = V_1$$

$$T_3 = \frac{V_3}{V_2} T_2 = r_c T_2$$

$r_c = V_3/V_2$, called the **cutoff ratio**

$$\frac{V_4}{V_3} = \frac{V_4}{V_2}\frac{V_2}{V_3} = \frac{V_1}{V_2}\frac{V_2}{V_3} = \frac{r}{r_c}$$

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{k-1} = r^{k-1}$$

$$\frac{T_4}{T_3} = \left(\frac{V_3}{V_4}\right)^{k-1} = \left(\frac{r_c}{r}\right)^{k-1}$$

$$\eta_{\text{th,Diesel}} = 1 - \frac{1}{r^{k-1}}\left[\frac{r_c^k - 1}{k(r_c - 1)}\right]$$

- *Thermal efficiency of the Diesel cycle increases with increasing compression ratio. Diesel cycle differs from the Otto cycle only by the term in brackets, which for is greater than unity. When the compression ratio is the same, the thermal efficiency of Diesel cycle is less than that of Otto cycle.*

$$\eta_{\text{th,Otto}} > \eta_{\text{th,Diesel}}$$

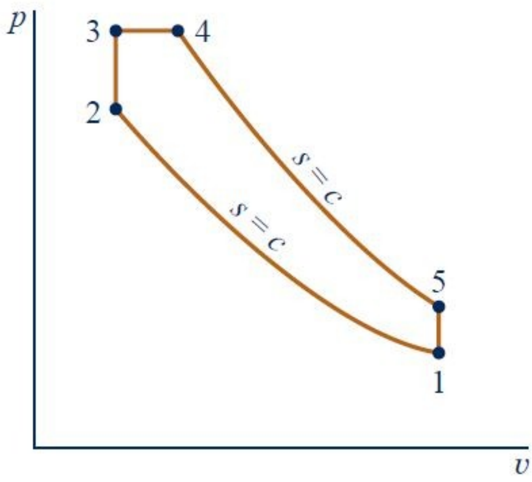
- *Diesel engines operate at much higher compression ratios and are usually more efficient than the spark-ignition engines. The diesel engines also burn the fuel more completely since they usually operate at lower revolutions per minute and the air–fuel mass ratio is much higher than spark-ignition engines. Thermal efficiencies of large diesel engines range from about 35 to 40 percent.*

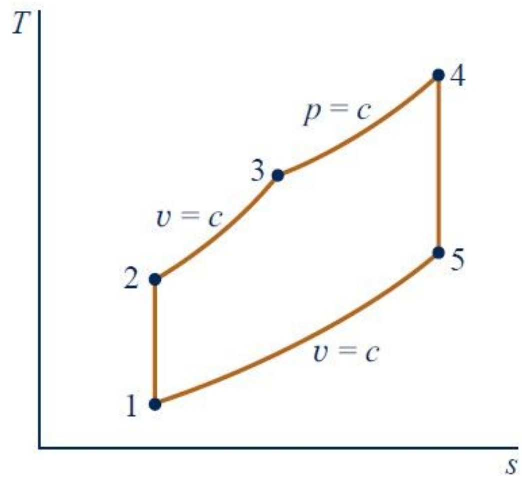
- *The higher efficiency and lower fuel costs of diesel engines make them attractive in applications requiring relatively large amounts of power, such as in locomotive engines, emergency power generation units, large ships, and heavy trucks.*

DUAL CYCLE

Dual cycle *models the combustion process as a combination of two heat-transfer processes, one at constant volume and the other at constant pressure. Both the Otto and the Diesel cycles can be obtained as special cases of the dual cycle. Approximating the combustion process in internal combustion engines as a constant-volume or a constant-pressure heat-addition process is overly simplistic and not quite realistic. Probably* ***dual cycle*** *is a better approach.*

Process 1–2: isentropic compression.
Process 2–3: constant volume heat addition.
Process 3–4: constant-pressure heat addition; also makes up the first part of the power stroke.
Process 4–5: isentropic expansion; the remainder of the power stroke.
Process 5–1: constant-volume heat rejection





<u>Cycle Analysis:</u>

$$\frac{W_{12}}{m} = u_2 - u_1 \qquad \frac{Q_{23}}{m} = u_3 - u_2$$

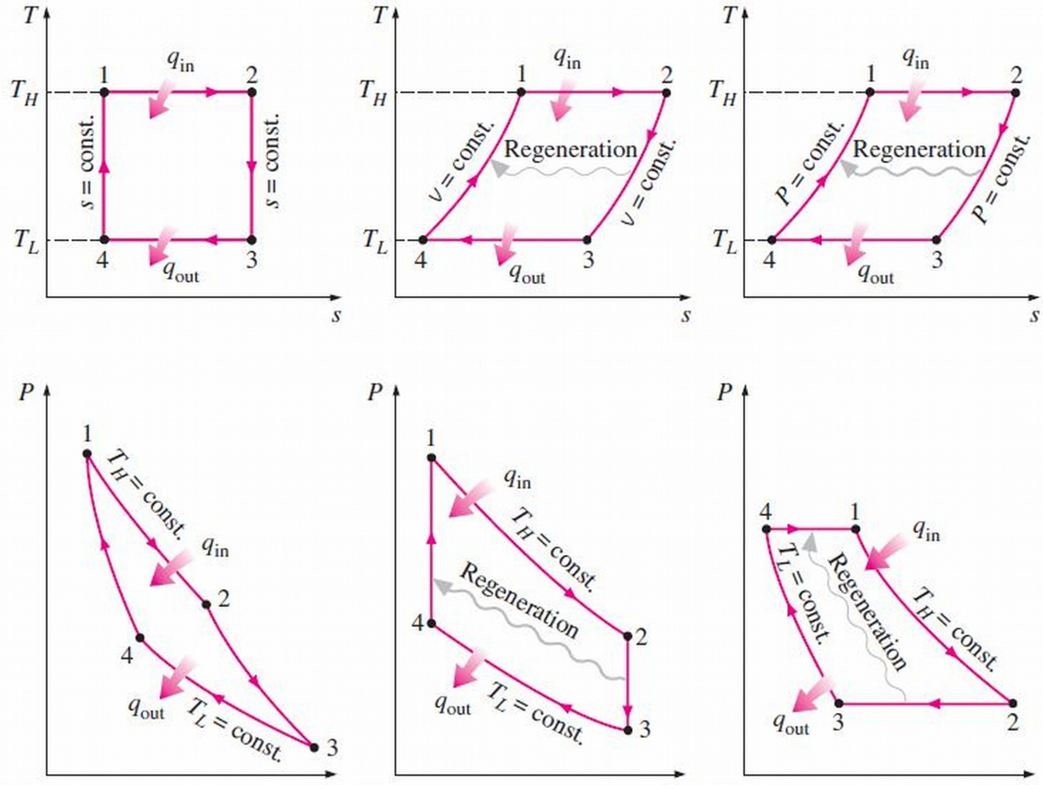
$$\frac{W_{34}}{m} = p(v_4 - v_3) \quad \text{and} \quad \frac{Q_{34}}{m} = h_4 - h_3$$

$$\frac{W_{45}}{m} = u_4 - u_5 \qquad \frac{Q_{51}}{m} = u_5 - u_1$$

$$\eta = \frac{W_{\text{cycle}}/m}{(Q_{23}/m + Q_{34}/m)} = 1 - \frac{Q_{51}/m}{(Q_{23}/m + Q_{34}/m)}$$

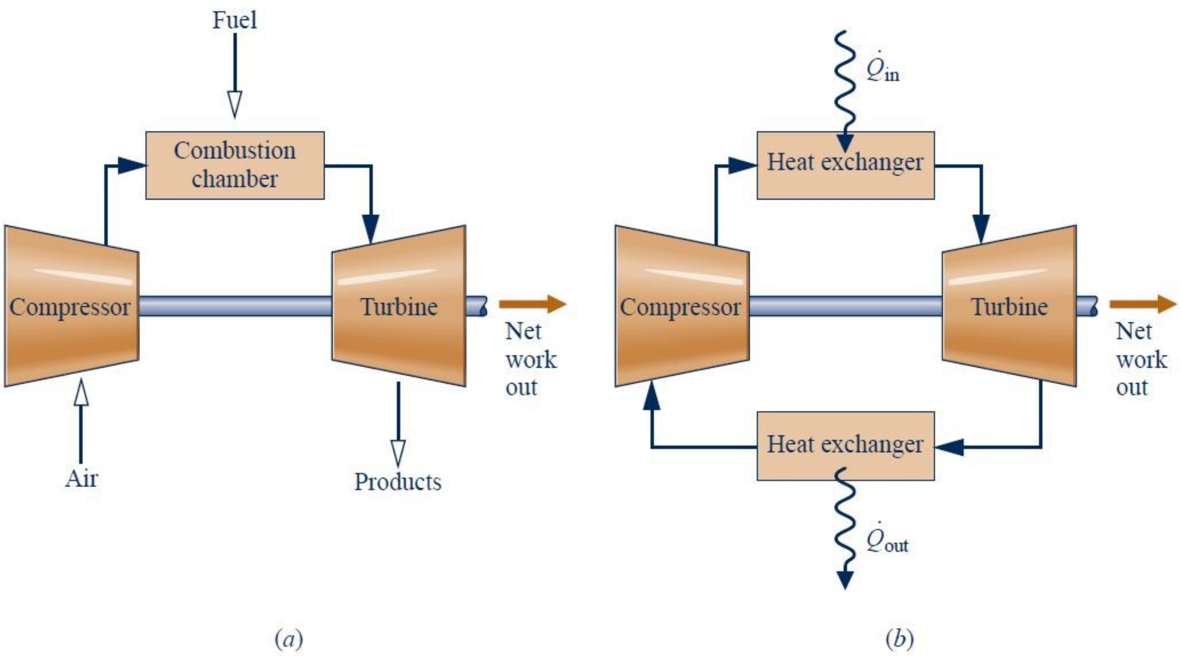
$$= 1 - \frac{(u_5 - u_1)}{(u_3 - u_2) + (h_4 - h_3)}$$

CARNOT, STIRLING AND ERICSSON CYCLE



(*a*) Carnot cycle (*b*) Stirling cycle (*c*) Ericsson cycle

INTRODUCTION TO GAS TURBINE ENGINES



OPEN AND CLOSED CYCLE GAS TURBINE

Gas turbines tend to be lighter and more compact than the vapor power plants. The favorable power- to-weight ratio of gas turbines makes them well suited for transportation applications (aircraft propulsion, marine power plants, and so on). Natural gas–fired gas has higher efficiencies, lower capital costs, shorter installation times, and better emission characteristics. The construction costs for gas-turbine power plants are roughly half that of comparable conventional fossil-fuel steam power plants.

Today's electric power-producing gas turbines are almost exclusively fueled by natural gas. However, depending on the application, other fuels can be used by gas turbines, including distillate fuel oil; propane; gases produced from landfills, sewage treatment plants, and animal waste and syngas (synthesis gas) obtained by gasification of coal.

Gas turbine power plants may operate on either an open or closed basis. The open mode is more common.

In an air-standard analysis two assumptions are always made:

- *The working fluid is air, which behaves as an ideal gas.*
- *The temperature rise that would be brought about by combustion is accomplished by a heat transfer from an external source.*

Working:

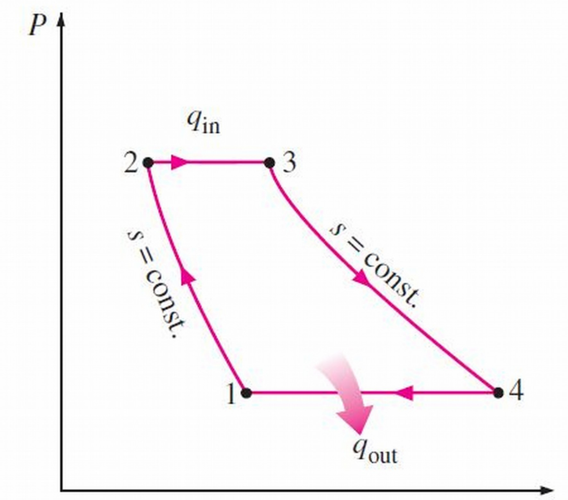
- *Atmospheric air is continuously drawn into the compressor, where it is compressed to a high pressure.*

- *Air then enters a combustion chamber, or combustor, where it is mixed with fuel and combustion occurs, resulting in combustion products at an elevated temperature.*

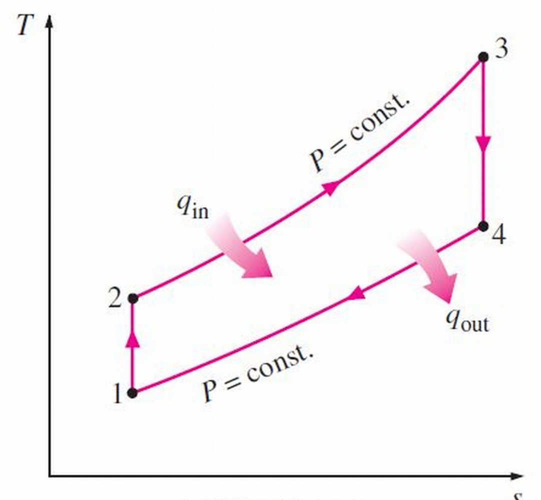
- *The combustion products expand through the turbine and are subsequently discharged to the surroundings. Part of the turbine work developed is used to drive the compressor; the remainder is available to generate electricity, to propel a vehicle, or for other purposes. The exhaust gases leaving the turbine are thrown out (not recirculated), causing the cycle to be classified as an open cycle*

BRAYTON CYCLE: THE IDEAL CYCLE FOR GAS-TURBINE ENGINES

Brayton cycle *is made up of four internally reversible processes:*
1-2 Isentropic compression (in a compressor)
2-3 Constant-pressure heat addition (in combustion chamber)
3-4 Isentropic expansion (in a turbine)4-1 Constant-pressure heat rejection



(b) P-v diagram



(a) T-s diagram

Cycle Analysis:

$$q_{\text{in}} = h_3 - h_2 = c_p(T_3 - T_2) \qquad q_{\text{out}} = h_4 - h_1 = c_p(T_4 - T_1)$$

$$\eta_{\text{th,Brayton}} = \frac{w_{\text{net}}}{q_{\text{in}}} = 1 - \frac{q_{\text{out}}}{q_{\text{in}}} = 1 - \frac{c_p(T_4 - T_1)}{c_p(T_3 - T_2)} = 1 - \frac{T_1(T_4/T_1 - 1)}{T_2(T_3/T_2 - 1)}$$

Processes 1-2 and 3-4 are isentropic, and $P_2 = P_3$ and $P_4 = P_1$.

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(k-1)/k} = \left(\frac{P_3}{P_4}\right)^{(k-1)/k} = \frac{T_3}{T_4} \qquad r_p = \frac{P_2}{P_1}$$

$$\eta_{\text{th,Brayton}} = 1 - \frac{1}{r_p^{(k-1)/k}}$$

The thermal efficiency of an ideal Brayton cycle depends on the pressure ratio of the gas turbine and the specific heat ratio of the working fluid. The thermal efficiency increases with both of these parameters. In most common designs, the pressure ratio of gas turbines ranges from about 11 to 16.

For a fixed turbine inlet temperature, the net work output per cycle increases with the pressure ratio, reaches a maximum, and then starts to decrease. There should be a compromise between the pressure ratio (thermal efficiency) and the net work output.

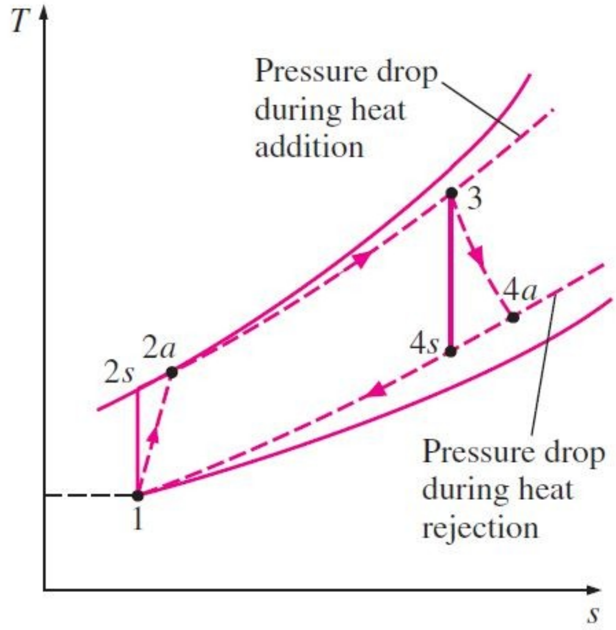
The fraction of the turbine work used to drive the compressor is called the back work. In gas-turbine power plants, the ratio of the compressor work to the turbine work, ***back work ratio,*** *is very high. Usually more than one-half of the turbine work output is used to drive the compressor. This is quite in contrast to steam power plants, where the back work ratio is only a few percent. Since a liquid is compressed in steam power plants instead of a gas, and the steady-flow work is proportional to the specific volume of the working fluid. Turbines used in gas-turbine power plants are larger than those used in steam power plants of the same net power output.*

The air in gas turbines performs two important functions: It supplies the necessary oxidant for the combustion of the fuel, and it serves as a coolant to keep the temperature of various components within safe limits. In gas turbines, an air–fuel mass ratio of 50 or above is not uncommon.

Irreversibilities And Losses:

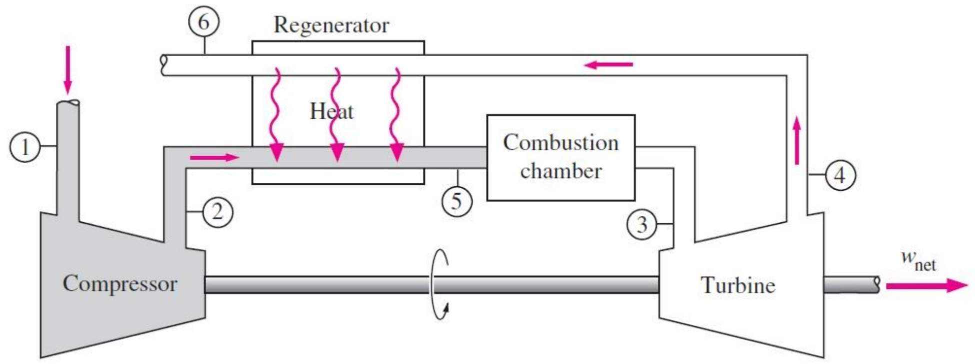
$$\eta_C = \frac{w_s}{w_a} \cong \frac{h_{2s} - h_1}{h_{2a} - h_1}$$

$$\eta_T = \frac{w_a}{w_s} \cong \frac{h_3 - h_{4a}}{h_3 - h_{4s}}$$



Some of irreversibilities and losses are: pressure drop during the heat-addition and heat rejection processes, the actual work input to the compressor is more, and the actual work output from the turbine is less because of frictional effects within the compressor and turbine, the working fluid would experience increases in specific entropy across these components. Most significant loss is combustion irreversibility.

<u>**BRAYTON CYCLE WITH REGENERATION**</u>



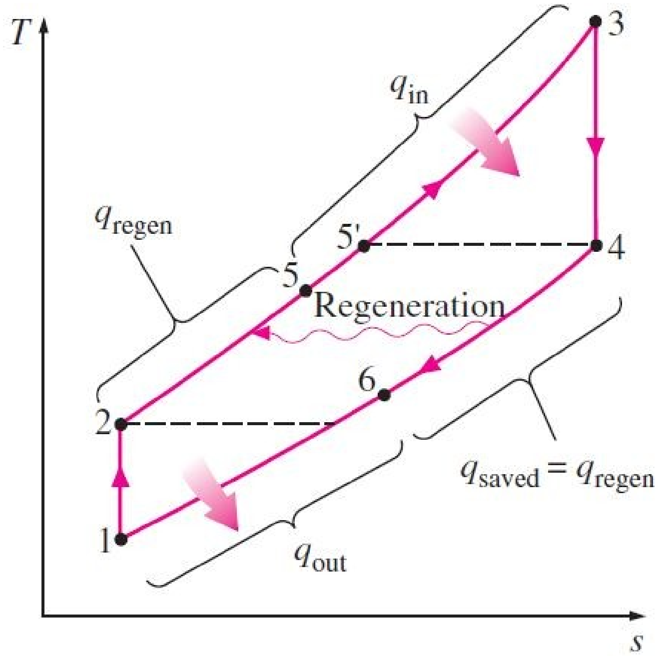
In gas-turbine engines, the temperature of the exhaust gas leaving the turbine is often considerably higher than the temperature of the air leaving the compressor. Therefore, the high-pressure air leaving the compressor can be heated by transferring heat from the hot exhaust gases in a counter-flow heat exchanger, known as a regenerator or a recuperator.

***Regenerator** allows the air exiting the compressor to be preheated before entering the combustor, thereby reducing the amount of fuel that must be burned in the combustor.*

Cycle Analysis:

$$q_{regen,act} = h_5 - h_2$$

$$q_{regen,max} = h_{5'} - h_2 = h_4 - h_2$$



*The extent to which a regenerator approaches an ideal regenerator is called the **effectiveness.** The effectiveness of most regenerators used in practice is below 0.85.*

$$\epsilon = \frac{q_{regen,act}}{q_{regen,max}} = \frac{h_5 - h_2}{h_4 - h_2}$$

When the cold-air-standard assumptions are utilized

$$\epsilon \cong \frac{T_5 - T_2}{T_4 - T_2}$$

The thermal efficiency of an ideal Brayton cycle with regeneration is:

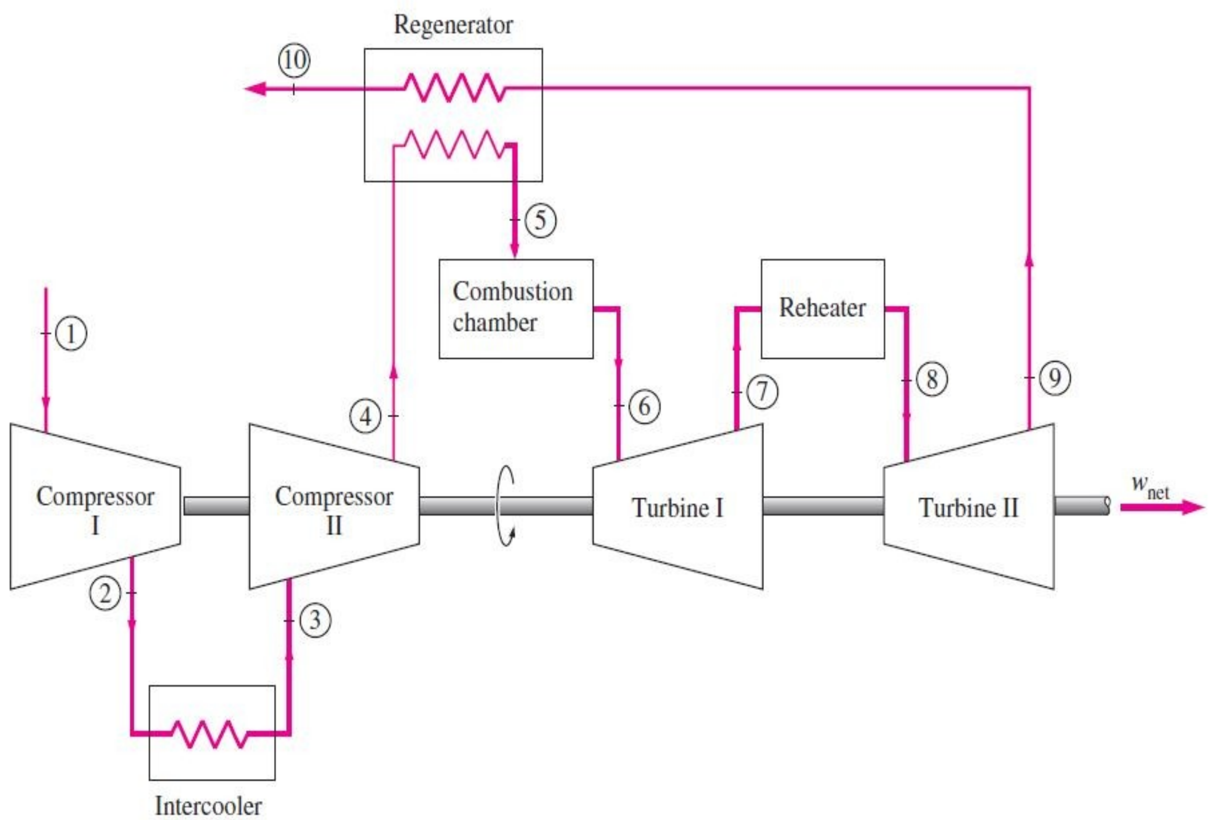
$$\eta_{th,regen} = 1 - \left(\frac{T_1}{T_3}\right)(r_p)^{(k-1)/k}$$

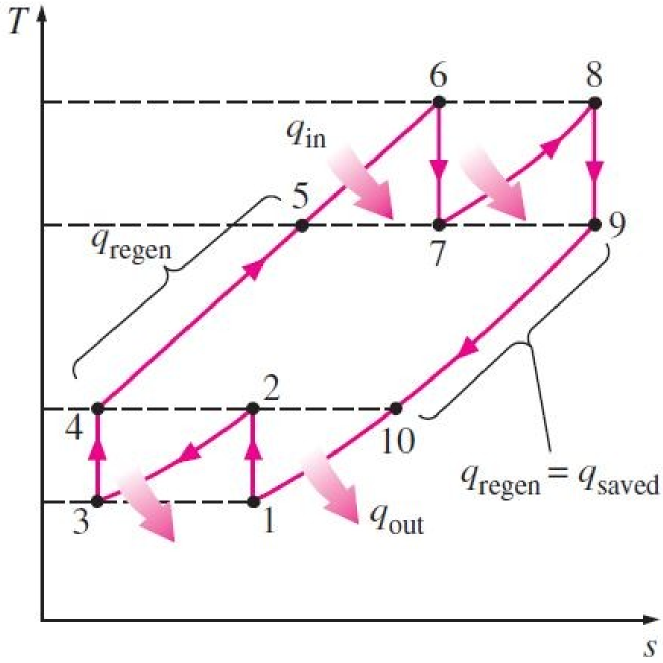
The thermal efficiency of the Brayton cycle increases as a result of regeneration since the portion of energy of the exhaust gases that is normally rejected to the surroundings is now used to preheat the air entering the combustion chamber.

The thermal efficiency of an ideal Brayton cycle with regeneration depends on the ratio of the minimum to maximum temperatures as well as the pressure ratio. Regeneration is most effective at lower pressure ratios and low minimum-to-maximum temperature ratios. The net work developed per unit of mass flow is not altered by the addition of a regenerator.

The use of a regenerator is recommended only when the turbine exhaust temperature is higher than the compressor exit temperature. Otherwise, heat will flow in the reverse direction (to the exhaust gases), decreasing the efficiency. This situation is encountered in gas-turbine engines operating at very high pressure ratios.

BRAYTON CYCLE WITH INTERCOOLING, REHEATING, AND REGENERATION





Cycle Analysis:

$$T_3 = T_1 \qquad T_5 = T_9 \qquad T_8 = T_6$$

Work input to a two-stage compressor is minimized when equal pressure ratios are maintained across each stage. This procedure also maximizes the turbine work output. For best performance.

$$\frac{P_2}{P_1} = \frac{P_4}{P_3} \quad \text{and} \quad \frac{P_6}{P_7} = \frac{P_8}{P_9}$$

The net work of a gas-turbine cycle is the difference between the turbine work output and the compressor work input, and it can be increased by either decreasing the compressor work or increasing the turbine work, or both. The work required to compress a gas between two specified pressures can be decreased by carrying out the compression process in stages and cooling the gas in between-(using multistage compression with intercooling). The work output of a turbine operating between two pressure levels can be increased by expanding the gas in stages and reheating it in between-(multistage expansion with reheating).
As the number of stages is increased the compression process becomes nearly isothermal at the compressor inlet temperature and the expansion process becomes nearly isothermal without raising the maximum temperature in the cycle.

The steady-flow compression or expansion work is proportional to the specific volume of the fluid. Therefore, the specific volume of the working fluid should be as low as possible during a compression process and as high as possible during an expansion process. This is precisely what inter-cooling and reheating accomplish.

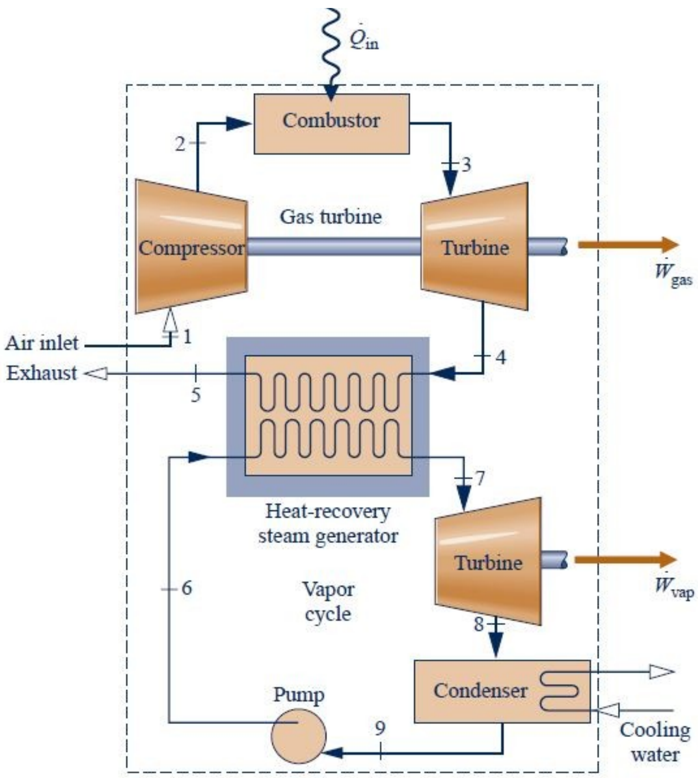
Reheating can be accomplished by simply spraying additional fuel into the exhaust gases between two expansion states because the exhaust gases are rich in oxygen.

The back work ratio of a gas-turbine cycle improves as a result of intercooling and reheating. However, this does not mean that the thermal efficiency also improves. The fact is, intercooling and reheating always decreases the thermal efficiency unless they are accompanied by regeneration. This is because intercooling decreases the average temperature at which heat is added, and reheating increases the average temperature at which heat is rejected.

COMBINED GAS TURBINE–VAPOR POWER CYCLE

A combined cycle couples two power cycles such that the energy discharged by heat transfer from one cycle is used partly or wholly as the heat input for the other cycle. The combined cycle of greatest interest is the gas-turbine (Brayton) cycle topping a steam turbine (Rankine) cycle.

The gas and vapor power cycles are combined using an interconnecting heat-recovery steam generator that serves as the boiler for the vapor power cycle. The combined cycle has the gas turbine's high average temperature of heat addition and the vapor cycle's low average temperature of heat rejection.



Cycle Analysis:

The thermal efficiency of the combined cycle:

$$\eta = \frac{\dot{W}_{\text{gas}} + \dot{W}_{\text{vap}}}{\dot{Q}_{\text{in}}}$$

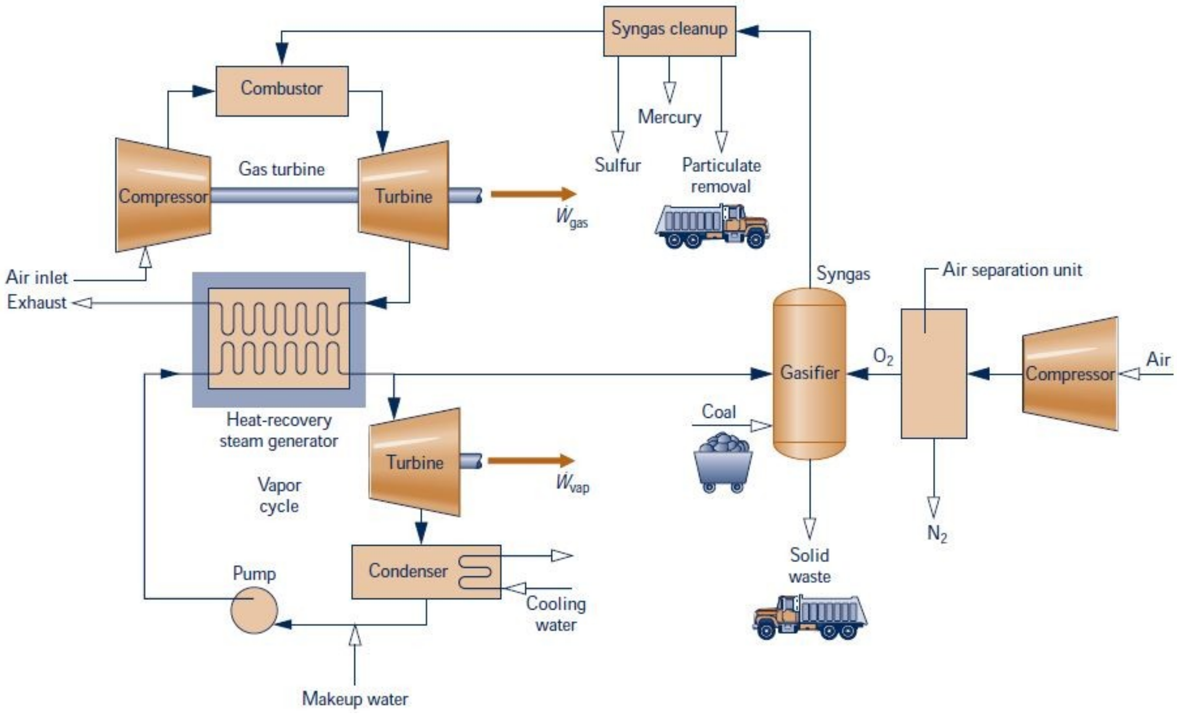
The relation for the energy transferred from the gas cycle to the vapor cycle for the system:

$$\dot{m}_{\text{v}}(h_7 - h_6) = \dot{m}_{\text{g}}(h_4 - h_5)$$

In steam cooling, relatively low-temperature steam generated in the companion vapor power plant is fed to channels in the blades of the high-temperature stages of the gas turbine, thereby cooling the blades while producing superheated steam for use in the vapor plant, adding to overall cycle efficiency.

H-class gas turbines also have single-crystal blades. Conventionally cast blades are polycrystalline. They consist of a multitude of small grains (crystals) with interfaces between the grains called grain boundaries. Adverse physical events such as corrosion and creep originating at grain boundaries greatly shorten blade life and impose limits on allowed turbine temperatures. Having no grain boundaries, single-crystal blades are far more durable and less prone to thermal degradation.

INTEGRATED GASIFICATION COMBINED-CYCLE POWER PLANTS



For decades vapor power plants fueled by coal have been the workhorses of electricity generation. However, human-health and environmental impact issues linked to coal combustion have placed this type of power generation under a cloud.

An IGCC power plant integrates a coal gasifier with a combined gas turbine–vapor power plant. Gasification is achieved through controlled combustion of coal with oxygen in the presence of steam to produce syngas (synthesis gas) and solid waste. Oxygen is provided to the gasifier by the companion air separation unit. Syngas exiting the gasifier is mainly composed of carbon monoxide and hydrogen. The syngas is cleaned of pollutants and then fired in the gas turbine combustor.

In IGCC plants, pollutants (sulfur compounds, mercury, and particulates) are removed before combustion when it is more effective to do so, rather than after combustion as in conventional coal-fueled power plants. IGCC plants emit fewer sulfur dioxide, nitric oxide, mercury, and particulate emissions than comparable conventional coal plants.